

University of Wollongong
Research Online

Faculty of Engineering and Information
Sciences - Papers: Part B

Faculty of Engineering and Information
Sciences

2019

Experimental investigation and performance analysis of an Organic Rankine Cycle for low-temperature heat to electricity generation

Lili Wei

Zhenjun Ma

University of Wollongong, zhenjun@uow.edu.au

Xuemei Gong

Xiujuan Guo

Follow this and additional works at: <https://ro.uow.edu.au/eispapers1>



Part of the [Engineering Commons](#), and the [Science and Technology Studies Commons](#)

Recommended Citation

Wei, Lili; Ma, Zhenjun; Gong, Xuemei; and Guo, Xiujuan, "Experimental investigation and performance analysis of an Organic Rankine Cycle for low-temperature heat to electricity generation" (2019). *Faculty of Engineering and Information Sciences - Papers: Part B*. 3673.
<https://ro.uow.edu.au/eispapers1/3673>

Research Online is the open access institutional repository for the University of Wollongong. For further information contact the UOW Library: research-pubs@uow.edu.au

Experimental investigation and performance analysis of an Organic Rankine Cycle for low-temperature heat to electricity generation

Abstract

© 2019 The Author(s) 2019. Published by Oxford University Press. This paper presents experimental investigation of low-temperature heat to electricity generation system based on Organic Rankine Cycle (ORC) using R152a as the working fluid. Both energy efficiency and exergy efficiency were analyzed based on the experiments. Although energy efficiency was low to 5.0% when the evaporating and cooling temperatures were 65°C and 11°C, respectively, the exergy efficiency reached 25%, which showed great competitiveness among low-temperature heat utilization technologies. To reveal the energy recovery proportion from the waste heat, both energy extraction efficiency and exergy extraction efficiency as well as energy and exergy loss paths were analyzed. When the heat source was 65°C, 14.9% of the maximum possible thermal energy in the heat source was absorbed by the organic working fluid, and 10.7% was transferred to the cooling medium. The power output contributed 0.64%. A total of 1.8% of the exergy in the heat stream flowed to the cooling medium. The start-up work takes dramatically 0.16% and 1.7% of energy and exergy, respectively. Other energy and exergy loss occurs due to the irreversibility of the heat transfer process and expansion process. Cascade ORC system could enlarge the temperature difference of the heat stream and raise the power output. However, the energy efficiency of the multi-stage ORC system is lower than single-stage system, since there was a downward trend of the temperature of heat source for the latter stage. ORC cycle can lower the temperature of heat source to 45°C.

Disciplines

Engineering | Science and Technology Studies

Publication Details

Wei, L., Ma, Z., Gong, X. & Guo, X. (2019). Experimental investigation and performance analysis of an Organic Rankine Cycle for low-temperature heat to electricity generation. *International Journal of Low-Carbon Technologies*, 14 (4), 500-507.

Experimental investigation and performance analysis of an Organic Rankine Cycle for low-temperature heat to electricity generation

Lili Wei^{1,*}, Zhenjun Ma², Xuemei Gong¹ and Xiujuan Guo¹

¹*School of Civil and Transportation Engineering, Ningbo University of Technology, Ningbo City, Zhejiang Province, China* ²*Sustainable Buildings Research Centre, University of Wollongong, 2522, Wollongong, New South Wales, Australia*

Abstract

This paper presents experimental investigation of low-temperature heat to electricity generation system based on Organic Rankine Cycle (ORC) using R152a as the working fluid. Both energy efficiency and exergy efficiency were analyzed based on the experiments. Although energy efficiency was low to 5.0% when the evaporating and cooling temperatures were 65°C and 11°C, respectively, the exergy efficiency reached 25%, which showed great competitiveness among low-temperature heat utilization technologies. To reveal the energy recovery proportion from the waste heat, both energy extraction efficiency and exergy extraction efficiency as well as energy and exergy loss paths were analyzed. When the heat source was 65°C, 14.9% of the maximum possible thermal energy in the heat source was absorbed by the organic working fluid, and 10.7% was transferred to the cooling medium. The power output contributed 0.64%. A total of 1.8% of the exergy in the heat stream flowed to the cooling medium. The start-up work takes dramatically 0.16% and 1.7% of energy and exergy, respectively. Other energy and exergy loss occurs due to the irreversibility of the heat transfer process and expansion process. Cascade ORC system could enlarge the temperature difference of the heat stream and raise the power output. However, the energy efficiency of the multi-stage ORC system is lower than single-stage system, since there was a downward trend of the temperature of heat source for the latter stage. ORC cycle can lower the temperature of heat source to 45°C.

Keywords: Organic Rankine Cycle; low-temperature thermal energy; energy efficiency; exergy efficiency

*Corresponding author.
weililinbut@outlook.com

Received 11 March 2019; revised 16 May 2019; editorial decision 16 May 2019; accepted 27 May 2019

1 INTRODUCTION

Converting low-grade heat into electric energy can effectively improve energy utilization efficiency and reduce environmental impacts. Organic Rankine Cycle (ORC) is an attractive option in low-grade heat conversion systems. Over the last two decades, many efforts have been devoted to working fluid selection, component optimization, system integration and adaptability to multiple heat sources.

The organic working fluid is one of the most important factors influencing the energy conversion efficiency. Drescher and Bruggemann [1] investigated suitable thermodynamic fluids in biomass plants, and the family of alkylbenzenes showed the highest efficiency. Mirzaei *et al.* [2] compared several working fluids in

waste heat recovery system of a metal smelting furnace, and m-xylene, P-xylene and Ethylbenzene had a higher net power output, a higher efficiency and a lower total cost in comparison to toluene, n-decane, benzene, dimethylcarbonate and cyclohexane. Glover *et al.* [3] investigated an automotive waste heat recovery system and reduced the number of potential working fluids from 105 to 16 through a combination of engineering judgement, legislation and health and safety concerns. Bahrami *et al.* [4] investigated a combined Stirling–ORC power cycle. The system efficiency was around 34–42% under the operating temperatures of 80–140°C. The performance of FC72, FC87, HFE7100, HFE7000, Novec649, n-pentane, n-decane, R245fa and toluene were tested, and the optimal candidates identified were toluene, HFE7100 and n-pentane. Guo *et al.* [5] investigated the integration of a

low-temperature geothermally powered ORC system with a commercial R134a-based heat pump system. The performance of this system using R152a, R134a, R600a, NH_3 , R236ea, R600, R145fa and R245ca were also investigated. It was found that R236ea and R245ca showed higher the ratio of power produced by the power consumed by the heat pump subsystem value (the ratio of power produced by the power generation subsystem to the power consumed by the heat pump subsystem).

Expander is another key component, which converts thermal energy into electricity. Pei *et al.* [6] manufactured a micro turbo-expander for an ORC system. The results showed that the rational speed of the expander can reach as high as 54 000 r/min and the system efficiency was 0.43. Pantano and Capata [7] carried out a comparison between volumetric expanders and an Inlet Forward Radial micro turbine for the exploitation of an onboard ORC energy recovery system. The screw motor was believed to be the most suitable expander type based on the compromise among efficiency, lubrication and reliability. It is commonly used in the small-to-medium capacity ORC system.

For the expander and transmission system, several parameters have influence on its performance. Tang *et al.* [8] developed mathematical methods and considered the expander speed, suction pressure and inlet superheat on the expander performance. The isentropic efficiency and volumetric efficiency decrease as the expander rational speed increases from 1250 to 6000 rpm, and they decrease as the suction pressure increases from 0.33 to 0.47. Pan *et al.* [9] investigated the regulation law of the turbine and generator, and the results indicate that there is a maximum value of transmission-generator efficiency with the variation of the rational speed.

Optimization also plays a key role in the efficiency improvement of ORC systems. Lu *et al.* [10] designed a 1-kW apparatus using scroll expander and tested eight working fluids. It was suggested that adding a recuperator to the ORC could raise the energy conversion efficiency. Braimakis and Karellas [11] optimized a double-stage ORC cycle in a waste heat conversion system and compared its performance with several working fluids. The results showed that the double-stage ORC can raise the exergetic efficiency by up to 25%, depending on the heat source temperature and the working fluid used. Wang *et al.* [12] compared five different types of ORC systems, including a simple ORC, ORC with an internal heat exchanger, ORC with an open feed organic fluid heater, ORC with a closed feed organic fluid heater and ORC with a reheater. The results showed that the ORC with an internal heat exchanger showed the best thermodynamic performance. Li [13] investigated an ORC-based solar thermal power system and compared it with a solar ORC with collector for direct vapor generation, a solar ORC with photovoltaic module and an osmosis-driven solar ORC. It was shown that the solar ORC with collector for direct vapor generation using two-stage collectors outperformed the others in terms of heat collection efficiency.

The performance evaluation is essential to understand their performance levels and cost effectiveness. Most commonly used criteria are energy efficiency and exergy efficiency. Bellos and Tzivanidis [14] designed a hybrid ORC driven by solar energy

and waste heat. The system efficiency varied from 11.6% to 19.7% when the temperature of the waste heat source changed from 150 to 300°C. Shokati *et al.* [15] recovered the waste heat from diesel engines using an ORC, and the heat efficiency was 8.85%. Zhang *et al.* [16] evaluated a low-grade energy conversion system using an energy analysis and a life cycle method, and R134a was chosen as the working fluid. It was found that the sustainability of the ORC system was less than that of wind, hydro and geothermal power plant but much greater than that of fossil fuel power plants. The energy proportion of the working fluid R134a accounted for 13.3% of the total input flows in the construction phase. Sun *et al.* [17] analyzed the exergy efficiency of an ORC cycle and an ORC-based combined cycle driven by low-temperature waste heat. The results showed that the exergy efficiency decreased with the increase of the evaporation temperature of the ORC [17]. Mirzaei *et al.* [2] and Braimakis and Karellas [11] also analyzed the exergy efficiency of ORC systems. The highest exergy destruction occurred in the condenser and economizer, while the pump led to a low exergy destruction. Among various working fluids, benzene showed the lowest exergy destruction.

Since exergy contained in the low-temperature heat sources is relatively low, exergy efficiency is commonly used to evaluate the performance of energy utilization systems in conjunction with energy efficiency. However, the paths of energy and exergy losses were not provided by the energy efficiency and exergy efficiency, and thus the systematic optimization may lack theoretical basis.

This paper presents the performance investigation of an ORC system. An experimental setup was developed, and its energy efficiency and exergy efficiency are analyzed. The energy and exergy flow paths were also analyzed to illustrate the causes of low energy efficiency and exergy efficiency.

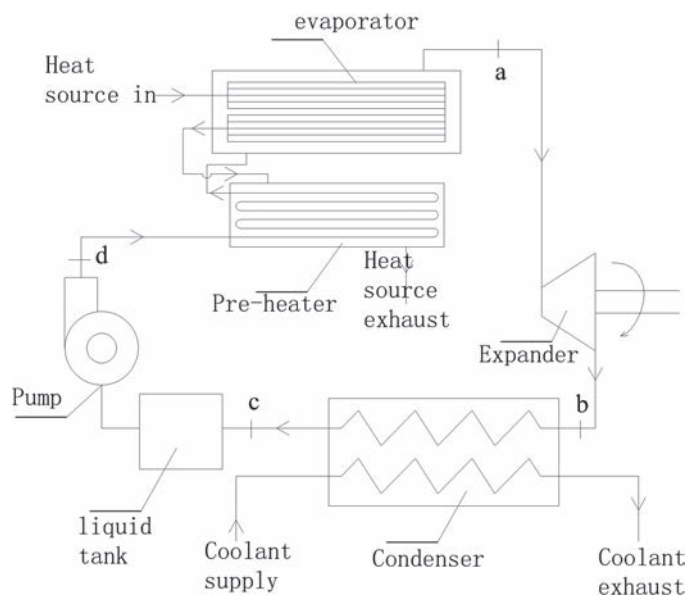


Figure 1. Illustration of an Organic Rankine Cycle.

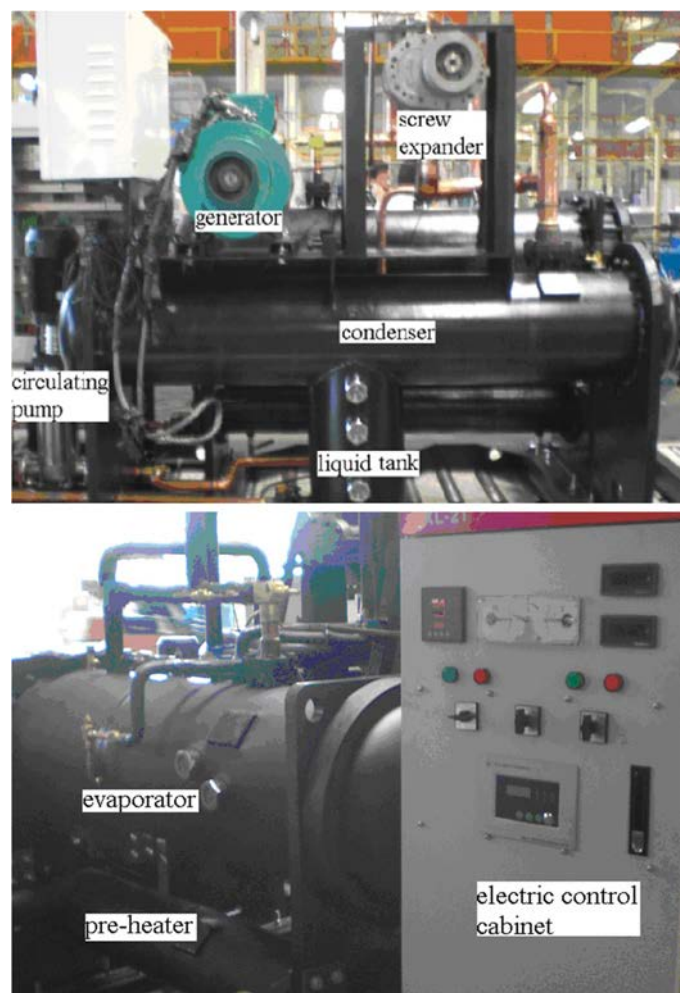


Figure 2. Schematic of the experimental apparatus.

1.1 Description of the experimental apparatus

The schematic of an ORC is shown in Figure 1 [18]. Screw expander was used because its working fluid can be superheated steam, saturated steam as well as wet steam, while turbine can only accept superheated or saturated steam as the working fluid.

The preheater and evaporator work as the heat exchangers to produce organic steam. The organic working fluid from the condenser is pumped to the preheater and is heated from sub-cooled liquid to saturated liquid. It is then further heated in the evaporator into the saturated steam. The heating process requires a long flow path, while the boiling process requires a large heat exchanging area and thus large heat transfer. Both the preheater and evaporator used are shell tube heat exchangers. The organic fluid flows in the tube side of the preheater and the shell side in the evaporator [19, 20].

Figure 2 shows the experimental apparatus used in this study. The refrigerant R152a (1,1-difluoroethane, CH_3CHF_2), whose ozone depletion potential is zero and global warming potential is as low as 0.023, was chosen as the working fluid.

1.2 Performance analysis

The heat source is the low-temperature hot water in order to simulate the use of natural low-temperature heat sources such as geothermal energy, solar energy and waste heat. Thus the average temperature of the heat source is described in Equation (1).

$$T_{h,m} = \frac{T_{h,in} + T_{h,out}}{2} \quad (1)$$

Hot water flows into the evaporator and transfers heat to the organic working fluid with a mass flow rate of m_h . The thermal energy exchanged (Q_h) is determined by Equation (2).

$$Q_h = \rho V_h \times c_p \times (T_{h,in} - T_{h,out}) \quad (2)$$

The environment temperature is T_0 , and the exergy transferred from the hot source to the organic fluid is calculated using Equation (3).

$$E_h = Q_h \left(1 - \frac{T_0}{T_{h,m}} \right) \quad (3)$$

The energy output is electricity W , which is the high-quality energy so its exergy is equal to energy W . Thus the energy efficiency η_{en} and exergy efficiency η_{ex} are determined as follows:

$$\eta_{en} = \frac{W}{Q_h} \quad (4)$$

$$\eta_{ex} = \frac{W}{E_h} \quad (5)$$

Energy efficiency and exergy efficiency were used as the performance indicators of the ORC cycle. However, the energy utilization efficiency of the heat source was not considered. For low-temperature thermal energy utilization technologies, the maximum extraction of heat from the heat source is the fundamental purpose. Energy extraction efficiency $\eta_{en,ext}$ and exergy extraction efficiency $\eta_{ex,ext}$ are defined as below:

$$\eta_{en,ext} = \frac{Q_h}{Q_{h,t}} \quad (6)$$

$$\eta_{ex,ext} = \frac{E_h}{E_{h,t}} \quad (7)$$

The overall energy efficiency and exergy efficiency of the energy conversion system are determined using Equations (8) and (9), respectively,

$$\eta_{en,sys} = \frac{W}{Q_{h,t}} \quad (8)$$

$$\eta_{ex,sys} = \frac{W}{E_{h,t}} \quad (9)$$

Table 1. Measurement range and accuracy of detection devices

Name	Type	Measurement range	Accuracy
Pressure transducer	—	−0.1–3 Mpa	1.0%
Temperature sensor	Pt100	−50–200°C	±0.3 + 0.005 t
Flow rate	Vapor Water	Vortex flowmeter Turbine flowmeter	1–10 m ³ /h 15–220 m ³ /h
Rotational speed	Non-contact digital tachometer	10–99 999.9 rpm	±1 rpm
Power meter		0–10 kw	±50 W

Table 2. Uncertainties of experimental parameters

Parameters	Measured			Calculated							
	T	V	W	Q_h	E_h	η_{en}	η_{ex}	$\eta_{en,ext}$	$\eta_{ex,ext}$	$\eta_{en,sys}$	$\eta_{ex,sys}$
Uncertainty (%)	0.62	1.0	0.5	1.44	1.01	1.9	1.53	2.92	2.04	1.97	1.53

2 RESULTS AND ANALYSIS

2.1 Uncertainty analysis

The data of temperature, pressure, volume flow rate of the heat stream and the organic fluid are monitored, and the interval is 10 s. The power capacity, voltage and current are also automatically recorded. All the sensors and their accuracy are presented in Table 1. The systematic and random errors for the efficiency can be calculated according to the standard [21]. The uncertainty $U(y)$ of y is composed of the uncertainty $U_A(y)$ and $U_B(y)$, which can be expressed as

$$U(y) = \sqrt{U_A^2(y) + U_B^2(y)} \quad (10)$$

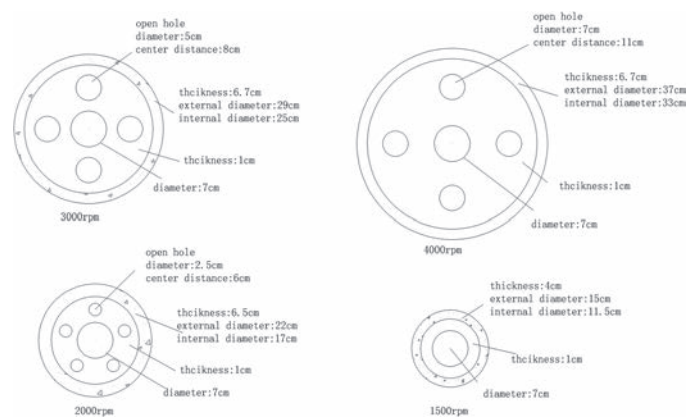
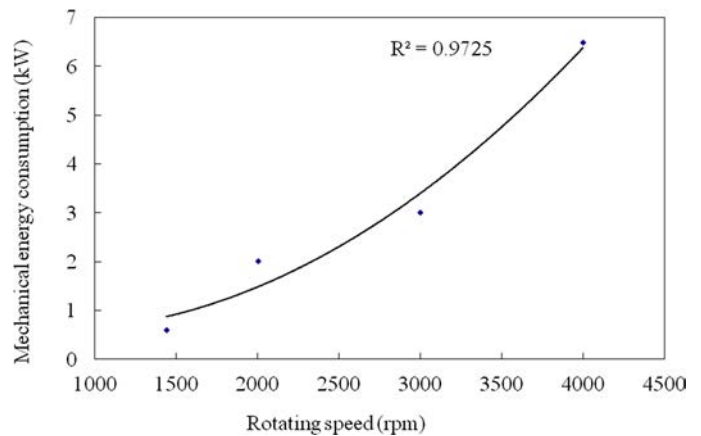
$$U_A(y) = \sqrt{\frac{\sum_{i=1}^n (y - \bar{y})^2}{n(n-1)}} \quad (11)$$

$$U_B(y) = \sqrt{\sum_{i=1}^n \left(\frac{\partial y}{\partial x_i} \right)^2 U_B^2(x_i)} \quad (12)$$

In the equations, y means the combined variance, such as Q_h and η_{en} ; x_i is the independent variable. The uncertainty results of the experimental results are presented in Table 2.

2.2 Start-up work

The screw expander drives the alternator through the belt drive. The rotational speed of the screw expander is determined by the diameter of the driving wheels of the alternator. A side view of the wheels is shown in Figure 3. They were made of cast iron, and opening holes were adopted to reduce the weight of the wheels. The circle in the middle was for the connection of the wheel with the shaft.

**Figure 3.** Side views of the wheels for 4000 rpm, 3000 rpm, 2000 rpm and 1500 rpm of the screw expander.**Figure 4.** Start-up work for typical rotational speed of the transmission system.

The rotation of the wheels consumes mechanical energy, which is needed to start the system. It equals to the kinetic energy when rotating, and the results are shown in Figure 4.

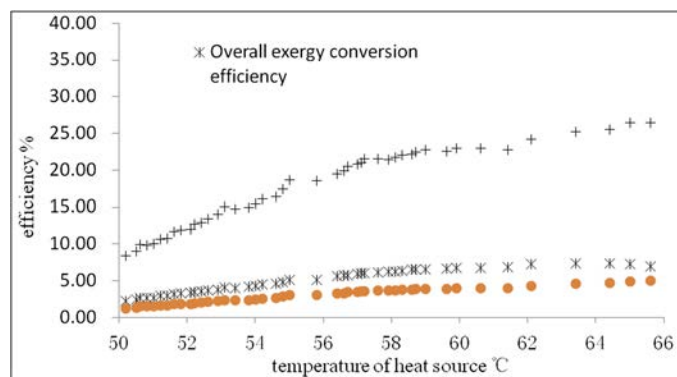


Figure 5. Efficiency analysis of R152a Rankine cycle.

2.3 Thermal and exergy efficiency analysis

Energy efficiency and exergy efficiency were defined in Section 3. Experiments were carried out when the cooling water was 11°C, the flow rate of the heat stream was 6.1 m³/s and the results are shown in Figure 5. Both energy efficiency and exergy efficiency were strongly correlated with the temperature of the heat sources. It demonstrated that the ORC system was not applicable when the temperature difference between the heat and cooling sources was lower than 35°C since the exergy efficiency was less than 8% and energy efficiency was lower than 2.3%. The energy efficiency of the experimental system was only 3.83% when the cooling water was 11°C and heat source was 65°C. However, the exergy efficiency was as high as 16%. During the experiments, the temperature of the heat source decreased from 65°C to 59°C. This means that the exhaust heat source could be used to drive another ORC cycle, and cascade ORC utilization of low-temperature energy could potentially facilitate the low-temperature energy recovery.

2.4 Energy and exergy extraction efficiency analysis

The energy and exergy extraction represent the energy absorbed from the heat source and the proportion of the useful energy to the total energy input, as shown in Figure 6. Both the energy extraction efficiency and exergy extraction efficiency increased with the increase of the temperature of the heat source. A higher energy or exergy extraction efficiency means less energy or exergy exhausted. The results showed that the necessity of using cascade ORCs to increase the energy conversion.

2.5 Energy and exergy flow analysis

The energy loss when the heat source was 65°C and is shown in Figure 7. Energy carried in the heat source was taken away mainly by the exhausted heat source stream, and the proportion of energy were 85.1%. The cooling water was another source, which took away 10.7% of energy. The final power output only accounted for 0.64% of the total energy supplied. The energy flow chart verified that raising the energy extraction efficiency is the most important way to improve the net energy conversion rate.

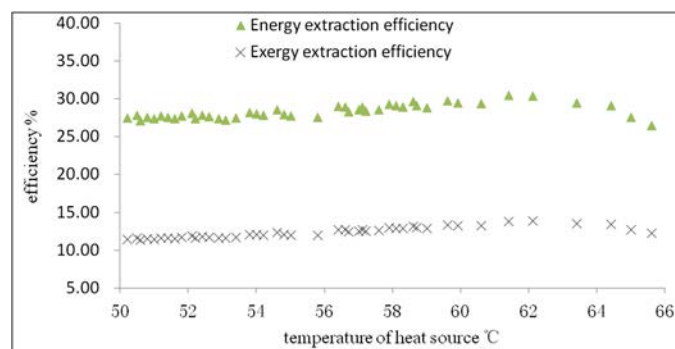


Figure 6. Energy extraction efficiency and exergy extraction efficiency.

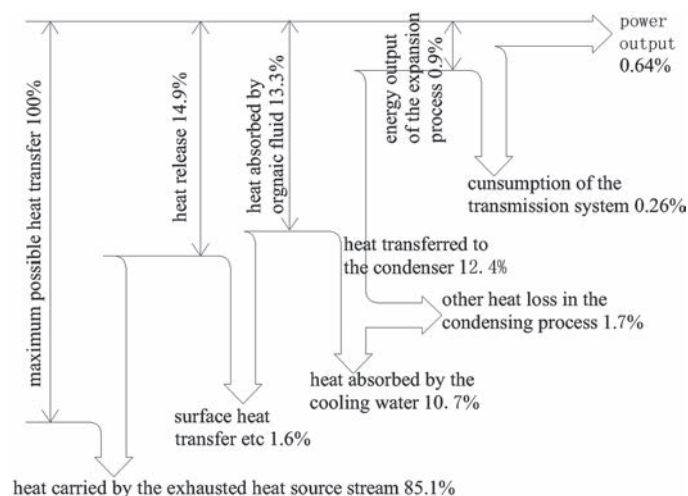


Figure 7. Energy flow of the low-temperature energy conversion system.

The exergy flow direction is similar to the energy flow direction, but the proportion was different, as shown in Figure 8. It is encouraging to see that the power output accounted for 7% of the maximum exergy transfer, and the exergy extraction efficiency was 25.2%. Although the cooling water had 10.7% of the maximum heat with only 2.0% of exergy, exergy loss in the exhaust heat source stream was decreased to 74.8%, which was still quite considerable.

The irreversible factors, such as temperature difference, led to 1.6% of energy loss and 6.6% of exergy loss. Therefore, further research on the heat transfer enhancement is needed to reduce this irreversible loss. The start-up work takes 0.16% and 1.7% of energy and exergy, respectively.

The energy and exergy flow paths were similar in direction but different in data. The reason was that the energy quality of thermal energy was relatively low and varied with the change of the temperature. Thus, the exergy efficiency was larger than the energy efficiency.

Energy loss and exergy loss during the expansion and electricity generation processes were also substantial. The performance of the expander should be optimized in order to achieve a higher isentropic efficiency.

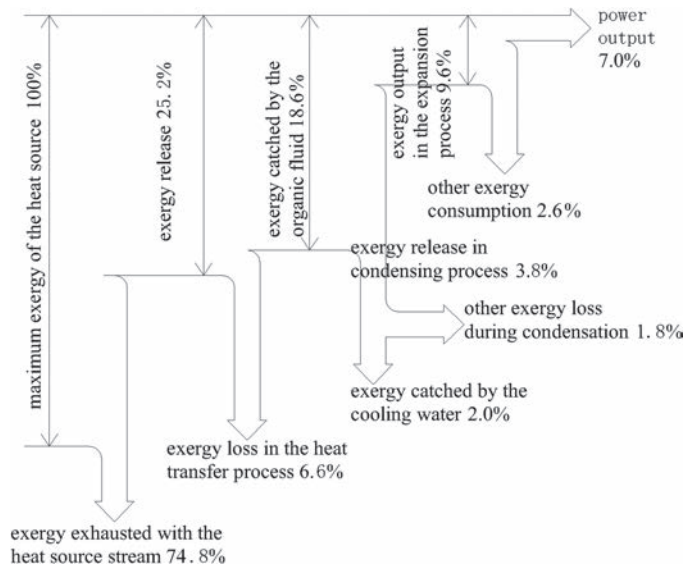


Figure 8. Exergy flow of the low-temperature energy conversion system.

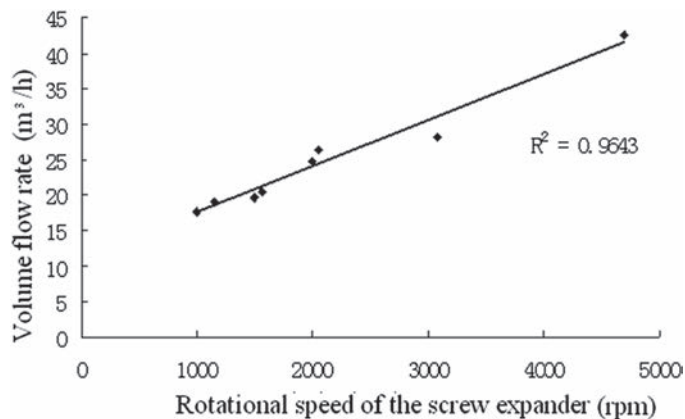


Figure 9. Relationship between the rotational speed of the expander and the flow rate of the steam.

3 PARAMETRIC STUDY AND PERFORMANCE IMPROVEMENT

3.1 Impact of the organic working fluid flow rate

For screw expander, the volumetric flow rate of the working fluid is proportional to the rotational speed. This was verified experimentally as illustrated in Figure 9. However, the linear trend did not pass through the coordinate origin due to the leakage between the two screws. The leakage was 11.08 m³/h and was relatively large.

Theoretically, a larger volumetric flow rate means more mechanical energy output. Figure 10 presents a comparison of the experimentally power output between the rotational speed of 2000 and 3000 rpm. A larger rotational speed means a larger volumetric flow rate of the working steam and more energy extracted from the heat source. The energy efficiency was higher when the rotational speed was 2000 rpm. Thus, there is an optimal rota-

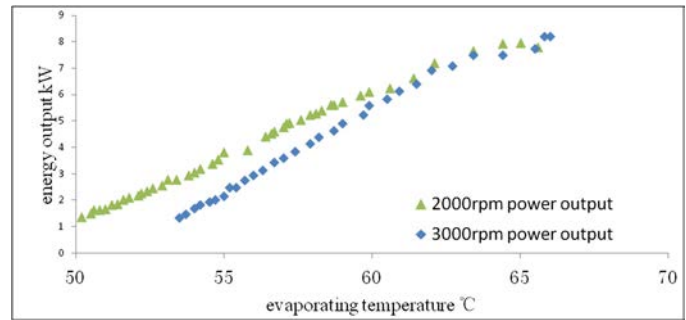


Figure 10. Power output at different rotational speed.

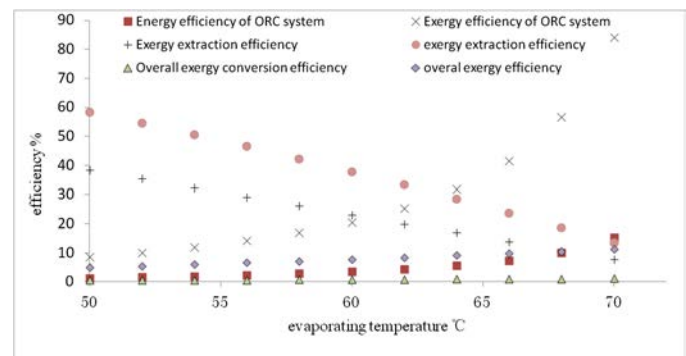


Figure 11. Efficiency analysis when the temperature of the heat source was 75°C, and the cooling water was 10°C.

tional speed that should be selected. All the experimental results were carried out based on 2000 rpm of the expander. This result is identical with the results of Tang *et al.* [8] and Pan *et al.* [9].

Shown in Figure 10, the power output increased as the evaporating temperature increased, but the slope was larger when the rotational speed was 3000 rpm. The energy input increased, and the volumetric flow rate of the organic steam also increased with the increase of the temperature of the heat source, and 3000 rpm performed better. Under a low evaporating temperature, the flow rate of the steam was not large enough to drive 3000 rpm expander, which showed the importance of optimization of the components in the system.

3.2 Impact of the exhausting temperature of the heat source

The energy and exergy loss flow charts showed the main points of systematic optimization. The energy extraction efficiency and exergy extraction efficiency were quite low, which limited the overall energy recovery efficiency. More exergy extracted means a larger temperature drop of the heat source, resulting in the decrease in the temperature of the organic saturated steam and the efficiency of the ORC cycle. Figure 11 demonstrates the theoretical energy efficiency and exergy efficiency of the ORC cycle. The energy extraction efficiency increased linearly with the decrease of the evaporating temperature when the heat source was maintained at 65°C. The variation in the exergy extraction efficiency

Table 3. Theoretical efficiency and experimental efficiency of the cascade ORC systems

		Exhaust temperature of heat source(°C)	Power output (kW)	η_{en} (%)	η_{ex} (%)	$\eta_{ex,ext}$ (%)
Single-stage ORC		58.9	7.81	5.02	32.37	11.09
Two-stage ORC	First ORC	58.9	7.81	5.02	32.37	11.09
	Second ORC	52.8	5.73	3.68	26.39	12.47
Combined operational performance			13.54	4.35	29.52	22.18
Three-stage ORC	First ORC	58.9	7.81	5.02	32.37	11.09
	Second ORC	52.8	5.73	3.68	26.98	12.47
	Third ORC	48	2.58	2.1	16.81	11.21
	Combined operational performance		16.12	3.72	26.36	30.91

was approximately linear with the temperature of the organic steam generated. However, the temperature and enthalpy of the organic steam decreased rapidly. The energy efficiency and exergy efficiency presented a downward trend. Considering the irreversible factors, the system efficiency was even lower than the theoretical efficiency. The maximum power output was obtained when the temperature drop of the heat source was 5–10°C.

3.3 Performance improvement using cascade ORC systems

To enlarge the temperature drop of the heat source, another choice is to use a cascade ORC system. The exhaust heat source of the first stage ORC cycle works as the inlet heat source of the second stage and so on. To simulate the cascade ORC system, three devices are connected in series to enlarge the temperature difference of the heat source. According to the experiments, the net power output, energy efficiency and exergy extraction efficiency of the two-stage and three-stage ORC systems are compared in Table 3. When the heat source was maintained at 65°C and cooling source was 11°C, the multi-stage ORC system had a larger net power output but offered lower energy efficiency and exergy efficiency, since there was a downward trend of the heat source of the latter stage. The overall exhaust stream should not be lower than 45°C in order to avoid too low energy efficiency and exergy efficiency. The increase in the power output was mainly due to the increase of the heat absorbed from the heat sources, which was manifested in the decrease in the exhaust temperature of the heat sources.

4 CONCLUSION

This paper investigated the experimental performance of an ORC with a low-temperature thermal energy conversion system using R152a as working fluid. Both energy efficiency and exergy efficiency were positively correlated with the evaporating temperature. The energy efficiency and exergy efficiency obtained from the experiment were 5.02% and 26.5% when the temperatures of the heat source and cooling source were 65°C and 11°C, respectively. Around 11% of energy and 25% of exergy of the heat source were extracted and converted into electricity, and the majority of them were exhausted.

The analysis of both energy flow path and exergy flow path showed that the energy loss and exergy loss occurred mainly in heat transfer process, the condensing process and the expansion process. The energy loss in heat transfer process and expansion process should be reduced through technology innovations and equipment performance improvement. The energy loss and exergy loss to the cooling medium were thermodynamically impossible to eliminate and can only be minimized. One possible solution was to use cascade ORC systems. Each stage of the ORC can lower the temperature of the heat source by 5–10°C, which means that the energy extraction efficiency can be improved by about 10%. The net power output can also be increased. Since the temperatures of the heat source for each stage had a downward trend, the energy efficiency and exergy efficiency showed the same downward trend.

The experimental investigation demonstrated the feasibility of ORC system for low-temperature thermal energy recovery. The energy and exergy flow analysis can provide insights on improvement of the system. To increase net power output, cascade ORC systems are a potential solution with enhanced efficiency.

NOMENCLATURE

T , temperature, °C; Q , heat transfer, kJ; V , volume flow rate, $\text{kg}\cdot\text{s}^{-1}$; ρ , density of the water, $\text{kg}\cdot\text{m}^{-3}$; c_p , Specific heat capacity, $\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{C}^{-1}$; E , exergy, kJ; η , efficiency, %; Subscripts, h , heat source; in , inlet of the ORC heat exchanger; out , outlet of the expander; 0 , surroundings; m , mean; en , energy; ex , exergy; ext , the exergy extraction from the heat source; t , total energy or exergy concatenated in the heat source stream; sys , systematic;

ACKNOWLEDGEMENT

The authors would like to thank the financial support granted from the National Natural Science Foundation of China (No. 51506102).

REFERENCES

- [1] Drescher U, Bruggemann D. Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plant. *Appl Therm Eng* 2007;27:223–8.

- [2] Mirzaei M, Ahmadi MH, Mobin M *et al.* Energy, exergy and economics analysis of an ORC working with several fluids and utilizes smelting furnace gases as heat source. *Thermal Science and Engineering Progress* 2018;**5**:230–7.
- [3] Glover S, Douglas R, Glover L *et al.* Automotive waste heat recovery: working fluid selection and related boundary conditions. *Int J Auto Tech* 2015;**16**:399–409.
- [4] Bahrami M, Hamidi AA, Porkhial S. Investigation of the effect of organic working fluids on thermodynamic performance of combined cycle Stirling–ORC. *International Journal of Energy and Environmental Engineering* 2013;**4**:12–20.
- [5] Guo T, Wang HX, Zhang SJ. Selection of working fluids for a novel low-temperature geothermally-powered ORC based cogeneration system. *Energy Convers Manag* 2011;**52**:2384–91.
- [6] Pei G, Li YZ, Li J *et al.* Performance evaluation of a micro-expander for application in low-temperature solar electricity generation. *J Zhejiang Univ Sci A* 2011;**12**:207–13.
- [7] Pantano F, Capata R. Expander selection for an on board ORC energy recovery system. *Energy* 2017;**141**:1084–96.
- [8] Tang H, Wu H, Wang X *et al.* Performance study of a twin-screw expander used in a geothermal organic rankine cycle power generator. *Energy* 2015;**90**:631–42.
- [9] Pan L, Wang H, Shi W. Regulation law of turbine and generator in Organic Rankine Cycle power generation experimental system. *Trans. Tianjin Univ.* 2014;**20**:237–42.
- [10] Lu Y, Roskilly AP, Jiang L *et al.* Analysis of a 1 kW organic Rankine cycle using a scroll expander for engine coolant and exhaust heat recovery. *Front Energy* 2017;**11**:527–34.
- [11] Braimakis K, Karellas S. Exergetic optimization of double stage Organic Rankine Cycle (ORC). *Energy* 2018;**149**:296–313.
- [12] Wang E, Zhang H, Fan B *et al.* Optimized performances comparison of organic rankine cycles for low grade waste heat recovery. *J Mech Sci Technol* 2012;**26**:2301–12.
- [13] Li J. Structural optimization and experimental investigation of the Organic Rankine Cycle for solar thermal power generation. *Ph.D. Thesis*. The University of Science and Technology of China, 2013.
- [14] Bellos E, Tzivanidis C. Investigation of a hybrid ORC driven by waste heat and solar energy. *Energy Convers Manag* 2018;**156**:427–39.
- [15] Shokati N, Mohammadkhani F, Farrokhi N *et al.* Thermodynamic and heat transfer analysis of heat recovery from engine test cell by Organic Rankine Cycle. *Heat Mass Transfer* 2014;**50**:1661–71.
- [16] Zhang H, Guan X, Ding Y *et al.* Emergy analysis of Organic Rankine Cycle (ORC) for waste heat power generation. *J Clean Prod* 2018;**183**:1207–15.
- [17] Sun W, Yue X, Wang Y. Exergy efficiency analysis of ORC (Organic Rankine Cycle) and ORC-based combined cycles driven by low-temperature waste heat. *Energy Convers Manag* 2017;**135**:63–73.
- [18] Gong X, Wei L, Guo X. Study of wet expansion in low-temperature heat-to-power conversion system. *Cryogenics* 2018; **222**(2): 62–68.
- [19] Wei L, Zhang Y, Mu Y. Study of evaporator in low-temperature energy conversion system using organic Rankine Cycle (ORC). *Cryogenics* 2015;**208**:40–5.
- [20] Wei L, Zhang Y, Mu Y *et al.* Design and research on low-temperature energy conversion system based on Organic Rankine Cycles (ORCs). *Acta Energetica Solaris Sinica* 2010;**33**:821–826, 2010.
- [21] State Bureau of Quality and Technology Supervision. Evaluation and presentation of uncertainty in measurement. China: Standards Press of China, 2012